

Evaluation of thermal efficiency and energy conversion of thermoacoustic Stirling engines

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ABSTRACT

Thermodynamic cycle transferring heat and work was executed in thermoacoustic engines, when the acoustic resonators substituted the moving mechanical components of the traditional heat engines. Based on the traveling-wave phasing and reversible heat transfer, thermoacoustic Stirling engines could achieve 70% of the Carnot efficiency theoretically, if the inevitable viscous dissipation in resonators was also counted as exported power. It should be pointed out an error on this efficiency evaluation in the previous literatures. More than 70% of the acoustic power production was often consumed by the side-branch resonator that was the essential configuration to build up a thermoacoustic Stirling engine. According to the simulation results and some experimental data, the actual available acoustic power consumed by the acoustic loads was restricted by the operating peak-to-mean pressure ratio, i.e. $|p_1/p_m|$. When the peak-to-mean pressure ratio operated on 4–6.5%, the thermal efficiency and power density of the available acoustic power reached higher levels. But the available acoustic power would approach zero when $|p_1/p_m|$ attained 10%. It was approved that the turbulence oscillation occurred on the higher $|p_1/p_m|$ (usually >4%) was the main reason of the excess dissipation in the side-branch resonator. This character of the available power limited the wide application of thermoacoustic Stirling engines. The evaluation of thermal efficiency and energy conversion also indicated the improving direction of thermoacoustic Stirling engines. Generators driven by the thermoacoustic Stirling engines were an effective way, due to the elimination of the side-branch resonator. To achieve a high power density and a high pressure ratio on the higher available power efficiency level, the standing-wave thermoacoustic engines might outvie the traveling-wave thermoacoustic engines. To enjoy the best features of standing-wave engines and traveling-wave engines simultaneously, exploiting multi-stage thermoacoustic engines, such as cascade engines, etc., would be an important research direction.

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1. Introduction

Acoustic energy could be converted into rotating shaft power, electricity and thermal energy, and vice versa. Thermoacoustic engine was a kind of device to realize the conversion between heat energy and acoustic power. Based on powerful oscillating pressure and oscillating flow along a temperature gradient in a pressurized fluid, thermoacoustic engine could pump heat from one place to another with high efficiency [1]. Thermoacoustic engines were derived from an interesting phenomena named as singing flame, which was the blare excited by the fuel combustion in an open-ended tube [2]. This kind of open-ended tube with inner combustion was also called as Rijke tube, which was often used to describe the combustion instability, occurred in the rocket combustion chamber [3]. Rijke was a professor of physics at the Leiden University in Netherlands, in 1859, he discovered a way of using heat to

sustain a sound in a cylindrical tube open at both ends [2]. The phasing between acoustic pressures and oscillating velocity in the heat source of Rijke tube was same as the phasing between pressure and velocity in a traveling-acoustic-wave. The idealized periodic cycle of the gas particles in Rijke tube was same as the Stirling cycle approximately, consisting of two constant-volume temperature changes and two constant-temperature volume changes. Stirling cycle was defined as a closed-cycle regenerative cycle, with the same high thermal efficiency as Carnot cycle. According to the intersection of the fields of thermodynamics and acoustics, Rijke tube was developed into the up-to-date traveling-wave thermoacoustic Stirling engines in the last several decades. The core component of thermoacoustic Stirling engines was regenerator units, sandwiched between two heat exchangers at different temperatures. Unlike the traditional heat engines, the phasing conditions to accomplish thermodynamic cycle were provided by acoustic circuits rather than moving mechanical components, which enhanced the reliability and life of thermoacoustic engines. The history of thermoacoustic engines started about

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1887, where Rayleigh discussed the possibility of pumping heat with sound [4]. Rayleigh defined a phenomenological criterion for the thermoacoustic coupling, which discussed the possibility of pumping heat with sound. Little further research occurred until Rott's work in 1969, who was the first to derive correct expressions for time-averaged energy transport in thermoacoustic coupling [1].

The most important development of the thermoacoustic technology was the invention of thermoacoustic Stirling heat engines, comprised of a traveling-wave loop circuit to accommodate regenerator units and a side-branch resonator to enhance regenerator impedance, shown as in Fig. 1, which attracted world-wide attention [5]. Based on the traveling-wave phasing and the reversible heat transfer, the thermodynamic process for the gas particles in the regenerator approximated to the efficient Stirling cycle. But it avoided the cranks, sliding seals or excess weight found in Stirling engines. The most efficient thermoacoustic Stirling engines, built in Los Alamos National Laboratory, achieved a relative Carnot efficiency of 41%, which was comparable with vapor compression systems today and automotive internal combustion engines [6]. Due to the high efficiency, thermoacoustic Stirling engines won a powerful competitive edge. Two decades before the invention of thermoacoustic Stirling engines, Ceperley had indicated the thermoacoustic conversion in a traveling-wave field was similar to the Stirling cycle [7]. But the high efficiency was overwhelmed by the low acoustic impedance of the working gas, which caused great viscous dissipation in the regenerator with high oscillation velocity amplitude [8]. The brilliant idea of thermoacoustic Stirling engines was to increase the local acoustic impedance of the regenerator in a loop using a long side-branch resonant tube. The relative Carnot efficiency of the regenerator units could attain 40–70% on different operating conditions, inferred from both the measured and calculated results [6–8].

But the actual available power efficiency of the thermoacoustic Stirling engine, used to drive a refrigerator, was much lower than the above mentioned [9]. How to evaluate the real efficiency of the thermoacoustic Stirling engines was also discovered in the deep research on other types of thermoacoustic engines [10–12]. To evaluate the real thermal efficiency, there were two different criterions: one based on the acoustic power exported into the bypass (included the side-branch resonator and acoustic load), and the other based on the consumed power by acoustic load. The latter criterion was significant in thermoacoustic refrigerators driven by thermoacoustic Stirling engines, because it represented the available power to drive refrigerators. It was few significative to evaluate thermal efficiency only based on the exported acoustic power from the traveling-wave loop. It should be pointed out the error on the evaluation on the thermal efficiency of thermoacoustic Stirling engines in previous literatures, which efficiency evaluation included the inevitable interior dissipation of the side-branch resonator. As an important but easy ignored issue, the side-branch resonant tube typically consumed more than 70% of the exported acoustic power from looped traveling-wave circuit. The dissipation

occurred in the side-branch resonator could be regarded as the transfer of the viscous loss that should occur in the regenerator if the side-branch resonator was eliminated. The irreversible loss of gas particle thermodynamic cycle and the resonator viscous dissipation both determined the available power efficiency of thermoacoustic engines. Therefore, to increase the reversibility of thermodynamic cycle should avoid accessional viscous losses as possible.

The famous first thermoacoustic Stirling heat engine achieved a thermal efficiency of 30% on the 700 °C heat source [6]. But the reported power included the available power in load and the dissipation in the side-branch resonator. Actually, the available thermal efficiency was only 24% due to 20% of the exported acoustic power was consumed in side-branch resonator, which dissipation was inevitable to ensure the regenerator high impedance. On the high acoustic intensity oscillation, the peak-to-mean pressure ratio amplitude (noted as $|p_1/p_m|$, where $|p_1|$ was the amplitude at antinodes) operating at 10%, the available thermal efficiency was only 6%, though the exported thermal efficiency was reported as high as 22%. More than 70% of the exported acoustic power was dissipated by the side-branch resonator, in order to keep this high intensity oscillation through the regenerator perfect thermodynamic cycle. The performance of all other thermoacoustic Stirling engines also testified the overrated thermal efficiency on the high intensity oscillation ($|p_1/p_m| \sim 10\%$) due to the inherent dissipation in external acoustic circuits [9,13]. The double loop thermoacoustic refrigerator with a shared side-branch resonator developed by Luo et al., the thermal efficiency to drive a refrigerator was only 7.18%, despite that the nominal exported power thermal efficiency was 24%. The recent study on a coaxial thermoacoustic Stirling cooler approved more than 50% acoustic power was dissipated in the resonator. The measured coefficient of performance relative to Carnot (COPR) of 25% was achieved, which nominal COPR in regenerator should be 45% [13].

As mentioned above, the inherent dissipation occurred in the side-branch resonator had been counted as one of exported power in the efficiency evaluation on thermoacoustic Stirling engines previously. On the high acoustic intensity oscillation ($|p_1/p_m| \sim 10\%$), the inherent dissipation often badly exceeded the available power. The acoustic circuits' dissipation was an inevitable cost to eliminate the moving mechanical parts, because the side-branch resonator was much greater than the regenerator units. The flowing state of the working fluid was concerned on the character of dissipation in acoustic circuits. This dissipation was a minor loss on the laminar flow oscillation, but would be a prominent loss on turbulence oscillation [14]. According to the side-branch resonator intended design, the regenerator impedance was commonly increased to 15–20 times of the special acoustic impedance of the working fluid. The velocity amplitude in the regenerator was often limited as 1–3 m/s. The pressure amplitude on the junction between the bypass and the loop was often 85% of the pressure antinodes amplitude, and the junction impedance was also com-

Illustrations

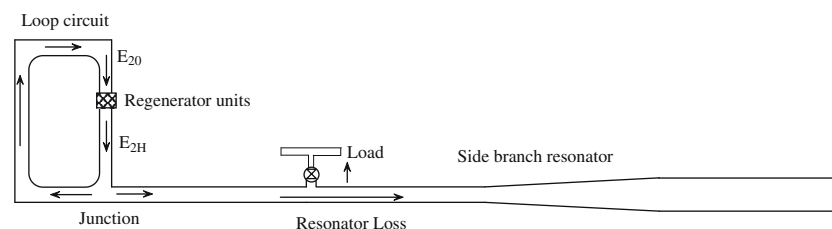


Fig. 1. The configuration of the thermoacoustic Stirling heat engine invented by Los Alamos National Laboratory, where arrowheads denoted the direction of the acoustic power flows.

parative to the gas special acoustic impedance. So the velocity amplitude on the junction was often kept as 10–60 m/s, which caused an important viscous dissipation in the side-branch resonator. Typically, the critical velocity amplitude of turbulence was only 17 m/s. Hence the dissipation would be enormous on high acoustic intensity oscillation.

Base on the next evaluation of thermal efficiency and energy conversion of thermoacoustic Stirling engines, it would indicate the way to improve efficiency. Thermoacoustic Stirling engines with side-branch resonator was not economical on the high pressure amplitude ($|p_1/p_m| \sim 10\%$), such as to drive a cryogenic pulse tube refrigerator. If the side-branch resonator was replaced by a mass-spring system in order to generate electricity, the dissipation would be decreased greatly. To solve the overmuch dissipation in the side-branch resonator, the compact thermoacoustic engines, such as multi-stage thermoacoustic engines and ultrasonic thermoacoustic engines, would be developed in future.

2. Resonant acoustical circuit impedance matching

It was a self-organizing process of energy during the thermoacoustic conversion process, so resonance was the main operating state. Similarly as the resonance occurred in radio-technology and electrician technology, acoustic resonance was the pure acoustic resistance behaving for a certain oscillation frequency in a non-source acoustic circuit's network containing acoustic inertia and compliance. The acoustic resonance occurred only on the zero system acoustic reactance, which determined the oscillating mode and the character of the sound field. The essential aim of the acoustic circuits design in thermoacoustic engines was to maintain a fitting resonance mode.

The acoustic circuit of thermoacoustic Stirling engines was composed of a main traveling-wave circuit and an assistant side-branch circuit to tune the acoustic impedance of the main circuit. In commonly, the main circuit used loop or coaxial bypass configurations, where was no essential difference between them. So the universality could be ensured only discussed either kind of the main circuits. The loop fashion was discussed in this paper [6]. As shown in Fig. 1, the thermoacoustic Stirling engine consisted of the thermoacoustic core (regenerator units, sandwiched between two heat exchangers at different temperatures) and acoustic circuits. And the acoustic circuits were composed of a loop (main circuit) and a side-branch resonator (exterior circuit). Essentially, the side-branch circuit was a $1/4$ -wavelength resonator. If the loop was cut from the junction with the side-branch resonator, it could boil down to a Rijke tube. Rijke tube was the simplest thermoacoustic hot air engine, converted heat into acoustic energy. The character of the sound field typically in the loop circuit was shown in Fig. 2, using helium as the working gas. The motion direction of the imaginary of velocity amplitude shifted just on top of the thermoacoustic core. Therefore, the phasing around the thermoacoustic core, which oscillating velocity leading to acoustic pressure, changed from near -90° to $+90^\circ$. Compromise between the viscous loss and irreversible loss in the regenerator units, the local traveling-wave region was often defined as the region where the absolute phase difference between the pressure oscillation and the velocity oscillation was less than 45° . Further more, the specific acoustic impedance in the so-called traveling-wave region was often more than 10 times of the working fluid special impedance, where the pressure amplitude neared the antinodes and the velocity amplitude neared the node. The traveling-wave phasing ensured thermodynamic cycle was close to ideal Stirling cycle. The high impedance ensured the velocity in regenerator was so low as to the viscous loss was reduced deeply. So the high impedance and near the traveling-wave phasing were both obtained in the

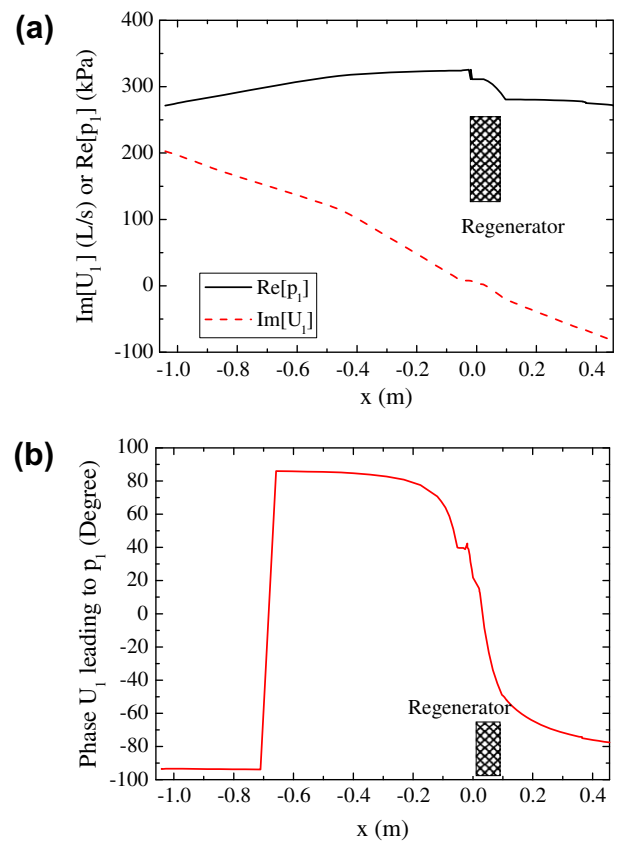


Fig. 2. (a) The sound field near the regenerator based on the simulation typically. (b) The confined local traveling-wave region near the regenerator units.

acoustic circuits of thermoacoustic Stirling engines. The location of the zero phase difference was called a sweet spot. The sweet spot in the loop circuit would move to the lesser velocity region, because of the asymmetric diffidence occurred on the junction. The sweet spot in the loop always existed in the regenerator due to the self-phase-tuning function of the regenerator along a temperature gradient. If the side-branch resonator was cut from the junction, the basic resonant frequency in the loop circuit corresponded to a $1/2$ -wavelength resonance according to the loop length. If only the loop was kept, the basic resonant frequency would be a full-wavelength resonance. The proportion of the local traveling-wave region length of wavelength was almost equivalent for different operating resonant frequency. So the absolute length of the local traveling-wave region increased when the operating resonant frequency decreased. To get the perfect thermodynamic cycle, the regenerator length should not exceed the local traveling-wave region length. The regenerator length determined the amount of the working fluid particles that accomplished the thermoacoustic conversion. So the acoustic intensity would be enhanced with the longer regenerator at the same temperature gradient. And the length of regenerator could also determine the heat conduction loss along the shell wall between heat exchangers at different temperatures. Therefore, to extend the absolute length of the regenerator was the most important technical requirement for an effective thermoacoustic engine, which was realized effectively by the side-branch acoustic circuit design in thermoacoustic Stirling engines.

As shown in Fig. 3, the simplified lumped impedance diagram of the thermoacoustic core was adopted to analysis the resonant frequency, where the footnote loop and resonator denoted the loop circuit and the side-branch resonator, respectively, Z denoted the acoustic impedance, R was acoustic resistance, L was

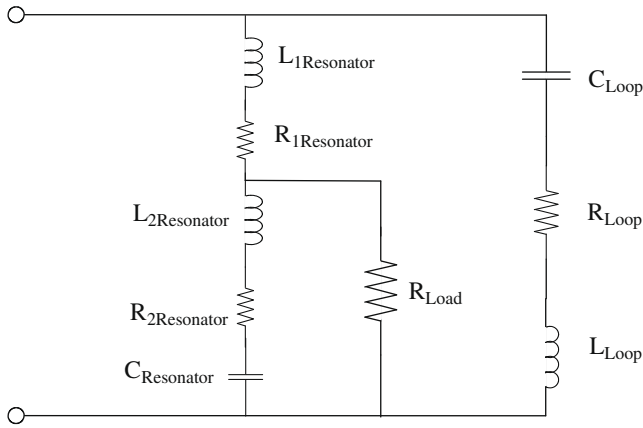


Fig. 3. The lumped-element impedance network of the thermoacoustic Stirling heat engine in Fig. 1.

the acoustic inertia and C was the acoustic compliance. All elements were much shorter than wavelength, so they could be modeled as lumped-elements. The length of the side-branch resonator was 2–3 times of the central axial length of the loop circuit. The resonant frequency of the side-branch resonator neared the $1/4$ -wavelength, which could be regarded as a standing-wave tube with one closed end and one opened end [15]. The resonant frequency of the complex circuit was determined as the formula (1) and (2) as below [5], where ω was the angle frequency and X was the acoustic reactance. The acoustic resistance had no effect on the resonant frequency except the influence on the onset conditions. The balance between the inertia and the compliance made the oscillation was easy excited by the minimal input of external energy.

$$|Z_{\text{Loop}}(\omega)|^2 X_{\text{Resonator}}(\omega) + |Z_{\text{Resonator}}(\omega)|^2 X_{\text{Loop}}(\omega) = 0 \quad (1)$$

$$\omega \approx \sqrt{1/L_{\text{Res}} C_{\text{Loop}}} \quad (2)$$

In the loop circuit, the acoustic compliance dominated on the upper 180° bend (just before thermoacoustic core), and the inertia dominated in the route from the junction to the compliance bend. Typically, the acoustic inertia was much lesser than the acoustic compliance in the loop, and the acoustic compliance was much lesser than the acoustic inertia in the side-branch resonator. So the resonant frequency could be evaluated as formula (2) approximately, determined primarily by the bypass reactance and the loop compliance. Commonly the resonant frequency of the complex resonators was close to the resonant frequency of the side-branch resonator. That meant the resonant frequency was equal to about $1/8$ – $1/12$ -wavelength resonance for the loop circuit. So the length of the local high impedance with near traveling-wave region was 8–12 times of that existed in a full-wavelength resonance loop.

Although the side-branch resonator was very effective to prolong the local traveling-wave region, it brought great additional viscous dissipation on high intensity oscillation. This additional dissipation of the side-branch resonator overwhelmed the high efficiency of the perfect thermodynamic cycle. According to next section, it was approved that the side-branch resonator was effective in lower amplitude oscillation, such as the peak-to-mean pressure ratio under 6.5%. The acoustic power flowed into the bypass was consumed by the load and the other was dissipated in the side-branch resonator. If the oscillating flow turned into turbulence, the dissipation in resonator got the run upon the available power by load. Strictly speaking, the efficiency calculated by the exported power from the junction was inaccurate, even in lower intensity

oscillation. The resonator dissipation could be regarded as the transfer of viscous loss which should occurred in the regenerator without the side-branch resonator. Except the exterior acoustic circuit, the compliance just before the regenerator and the regenerator self had the function to tune the position of the sweet spot, which would be discussed in other articles.

According to the sound field, the acoustic inertia of the loop before the thermoacoustic core could be ignored due to the minimum velocity. And the acoustic compliance of the junction could be ignored due to the minimum acoustic pressure. The operating frequency could be reduced if the compliance of the loop or the junction inertia increased. The function of the side-branch resonator was just to increase the junction inertia, which extended the traveling-wave region with high impedance. The resonator inertia was determined by the aspect ratio of the resonant tube, which was expressed as formula (3), where r was the radius of the tube, l was its length and ρ_m was the mean density of the working fluid. The diameter of the resonant tube was often greater than the viscous penetration depth, so the acoustic resistance on the internal surface of the tube could be expressed as formula (4) according to boundary approximation, where μ was the dynamic viscosity and δ_v was the viscous penetration depth. In ordinary, the acoustic resistance was much less than the inertia because $\delta_v \ll r$. As an easy concealed issue, the analogy between the acoustic resistance and the resistance in electric circuits was unfit in the turbulence acoustic oscillation, when the acoustic resistance consumed power was in direct proportion with the velocity cube.

$$X = \frac{\omega \rho_m l}{\pi r^2} \quad (3)$$

$$R = \frac{l}{\pi r^3} \sqrt{2\mu\omega\rho_m} = \frac{\omega\rho_m l}{\pi r^3} \delta_v = \frac{\omega\rho_m l}{\pi r^2} \frac{\delta_v}{r} \quad (4)$$

The diffidence of the oscillating volume velocities on the junction could be treated as the interference between two traveling-waves in opposite directions within $1/4$ -period lag in time, as shown in Fig. 4 typically. Because of the resistance of the side-branch resonator and the load, the net acoustic power propagated into the bypass circuit. And the acoustic power propagated in the loop circuit was nearly five times of the net power exported into the bypass. The acoustic powers propagated along the bypass in two opposite directions were much greater than that propagated along the traveling-wave loop. The average oscillating velocity in the bypass was greater than that in the loop, as often induced turbulence oscillation in the bypass. The net acoustic power transmitted into the bypass was mainly consumed by the resonator inner surface, due to the relative greater load resistance. The load resistance was much greater than the resonator inertia, so the load shared less volume velocity that carried acoustic power. In a laminar flow oscillation, the load consumed greater power than the resonator, because of the relative great load resistance with minor velocity. In a turbulence oscillation, the resonator resistance increased about $\langle u_1 \rangle$ times. So much lesser volume velocity shared by the load than that shared by the resonator. Therefore, the acoustic power distributive proportion between the load and the resonator was different for the laminar flow oscillation and the turbulence oscillation. In conclusion, the turbulence decreased the relative quantity of the available power (consumed by the load) on a high peak-to-mean pressure ratio oscillation, which caused a great error on the efficiency evaluation if all the power exported into the bypass was counted.

The different circuit impedance was also compared typically, which shown the side-branch reactance was much lesser than the load resistance. Typically, the load resistance was 9 – 28 MPa s/m^3 , and the compliance impedance in the loop was about 3.5 MPa s/m^3 , the inertia of the side-branch resonator was about 1.5 MPa s/m^3 . The calculated junction volume velocities ratio of the bypass to loop was 2.33 approximately, and the measured

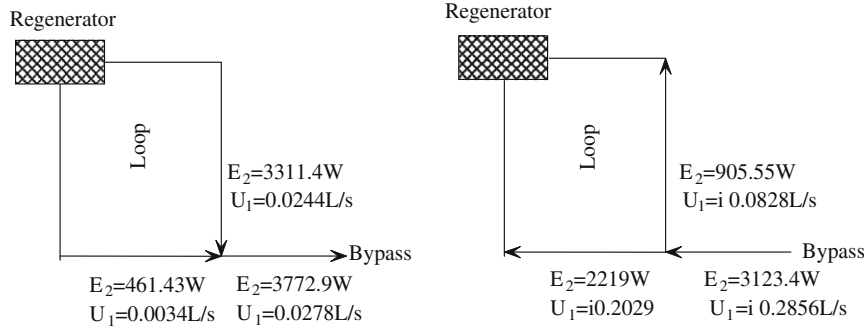


Fig. 4. The acoustic power and the volume velocity flow on the junction were treated as the superposition of two opposite traveling-waves.

value in experiments was 2.46 [4]. More loads using the parallel connection could increase the available power. Turbulence would be turned into laminar flow if the loads were enough.

3. Acoustic power losses on different $|p_1/p_m|$

As the above mentioned, the laminar flow or the turbulence induced different available power efficiency, which flow state was depended on the peak-to-mean pressure ratio. In this section, the transmission and the loss of the acoustic power in different elements were analyzed, respectively.

Firstly, the acoustic power was produced in the regenerator similarly a diode amplifier, which gain factor of the oscillating volume velocity (g) was expressed in formula (5), where f was the thermoacoustic function averaged in space, and the footnotes k and v denoted thermal and viscous, dT_m/dx was the mean temperature gradient in the regenerator, and Pr was the Prandtl number of working fluids [14]. Because the pressure had few changes in the regenerator units, the acoustic power gain was amplified as formula (6) expressed [14]. It was evident that the power gain was proportional to the acoustic power on the entrance of the regenerator (E_{20}) and the temperature gradient dT_m/dx .

$$g = \frac{f_k - f_v}{(1 - f_v)(1 - Pr)} \frac{1}{T_m} \frac{dT_m}{dx} \quad (5)$$

$$dE_2 = \frac{1}{T_m} \frac{dT_m}{dx} E_{20} \text{Re}[f_k] + \frac{1}{2} \frac{1}{T_m} \frac{dT_m}{dx} \text{Im}[\tilde{p}_1 U_1] \text{Im}[-f_k] \quad (6)$$

In the lossless oscillation with the flow channel that hydrodynamic radius was much less than thermal penetration depth (meant $f_k \approx 1$), the ideal amplified gain was determined by the temperature gradient, shown as formula (7) and (8), deduced from formula (6). The absolute temperature ratio of the hot end to cold end of the regenerator (T_H/T_0) was just the acoustic power ratio of the entrance to exit of the regenerator (E_{2H}/E_{20}). Ordinarily, T_H/T_0 was about 2–3, which meant the acoustic power exported into bypass was 1–two times of that cycled along the loop circuit.

$$\frac{dE_2}{dx} = \frac{1}{T_m} \frac{dT_m}{dx} E_{20} \quad (7)$$

$$E_{2H}/E_{20} = T_H/T_0 \quad (8)$$

Considered the viscous loss and thermal relaxation loss occurred in the regenerator, the acoustic power could be calculated using formula (9), where r_v and r_k were the acoustic resistance per unit length caused by viscous loss and thermal relaxation, respectively. The gain was proportional to volume velocity $|U_1|$, but the viscous loss was proportional to the square of volume velocity. Therefore, the higher power gain must reduce the efficiency. Compromise between the power level and the efficiency level (based on the acoustic power exported into bypass), the appropriate oscillating velocity

amplitude averaged in the regenerator should be 2–3 m/s according the simulation and some experimental results, as shown in Fig. 5. The actual losses in regenerator were calculated using formula (10) to simulate.

$$\frac{dE_2}{dx} = -\frac{r_v}{2} |U_1|^2 - \frac{1}{2r_k} |p_1|^2 + \frac{1}{2} \text{Re}[g \tilde{p}_1 U_1] \quad (9)$$

$$\Delta E_{2\text{Loss}} = E_{20} \frac{T_H - T_0}{T_0} - E_{2H} \quad (10)$$

The normalized acoustic power gain per unit length of the regenerator was defined by Ceperley, which was influenced by some dimensionless parameters such as the square of acoustic impedance, the operating frequency and the temperature gradient [8]. According to Ceperley's analytical formula, 79% of Carnot efficiency

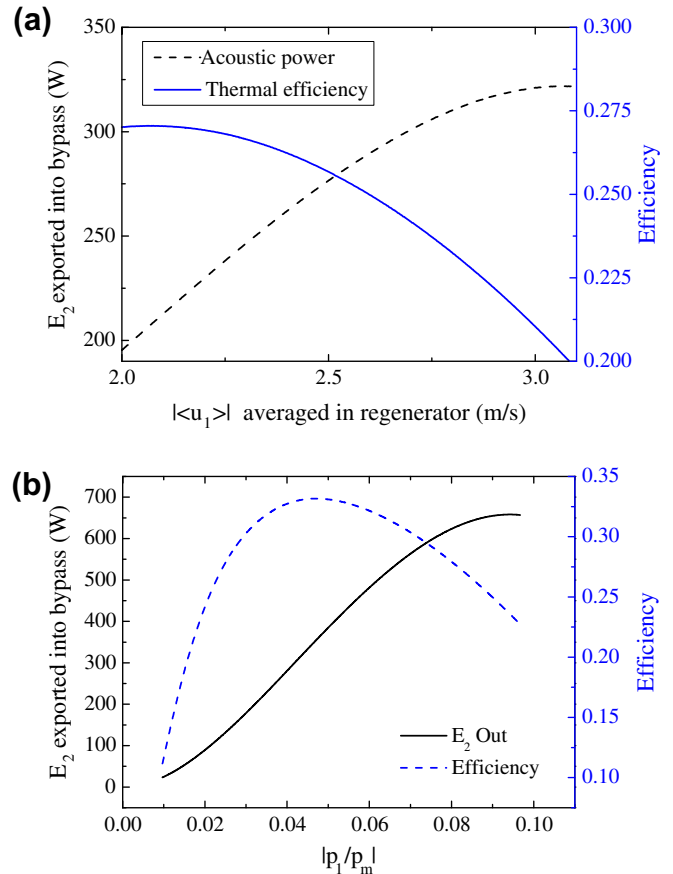


Fig. 5. (a) Compromise of the junction power and the efficiency on different velocity amplitudes. (b) Compromise of the power and the efficiency on different pressure ratios.

could be acquired, if the local impedance of the regenerator was multiplied by 10. The effective methods to increase impedance of the regenerator were the most important technique, such as the attempt to add a side-branch resonator. But the side-branch resonator and the loop circuit were both consumed acoustic power. These dissipation included the friction loss along the circuits and the local resistance loss at the break interfaces among the different elements. All the losses varied with different velocity amplitude. The available power, the power gain minus all kinds of losses, was hence the function of velocity amplitude.

The oscillation velocity amplitude was difficult to measure in a high charging pressure system using laser doppler velocimeter (LDV), etc. [16]. But the velocity amplitude was easy scaled by the peak-to-mean pressure ratio, $|p_i/p_m|$. With ideal-gas assumption, shown as formula (11), the peak-to-mean pressure ratio essentially was the velocity amplitude multiplied by a factor $N\gamma/c_0$, where N was the dimensionless impedance defined as the acoustic impedance divided by the special impedance of the working fluids, γ was the ratio of isobaric to isochoric specific heats, c_0 was the sound speed and $\langle u_1 \rangle$ was the velocity amplitude spatial average perpendicular to the axial. Noticed that the dimensionless local impedance N would be only multiplied 1.8 when $|p_i/p_m|$ was magnified 10 times typically, shown as in Fig. 6a. According to Fig. 6b, $|p_i/p_m|$ could reflect the velocity amplitude, which determined the available power and the actual efficiency. According to the pressure–volume diagram, the net area of a particle's thermodynamic cycle in the regenerator was proportional to the oscillation displacement amplitude, which was the velocity amplitude divided by the angle frequency. Considered the viscous loss in

the regenerator, the loop and the regenerator, the available power efficiency was not linear growth or drop with the rising $|p_i/p_m|$.

$$\frac{p_i}{p_m} = \frac{p_i}{\rho_m RT} = \frac{p_i \gamma}{\rho_m c_0^2} = \frac{p_i}{N \rho_m c_0} \frac{N \gamma}{c_0} = \frac{N \gamma}{c_0} \langle u_1 \rangle \quad (11)$$

The peak-to-mean pressure ratio was an important parameter in driving a pulse tube cryogenic refrigerator or a generator, which value was often required more than 10%. The measurement in a 165.8 dB ($p_i/p_m = 4\%$) intensity sound field approved the enormous turbulence of Rayleigh streaming [17]. More non-linear phenomena was confirmed when $|p_i/p_m|$ exceeded 10%, which should caused a bad dissipation [14]. Therefore, high $|p_i/p_m|$ was not the pursued goal in Los Alamos National Laboratory, although they had achieved 16% of $|p_i/p_m|$ in 1992 [18]. The famous first thermoacoustic Stirling engine got the highest available thermal efficiency (24%) at the 6.1% of $|p_i/p_m|$, but the available power efficiency was only 6% at the 10% of $|p_i/p_m|$, due to the serious dissipation in the side-branch resonator. To drive a cryogenic pulse tube refrigerator ($T_c < 120$ K), thermoacoustic Stirling engines might be not worth the candle.

The other method to enhance the local impedance in the regenerator, such as what Ceperley imagined, used an extended cross section area in regenerator. But its influence was fainter than the function of the exterior circuit. For example, the simulation results, as shown in Figs. 7 and 8, the local averaged impedance was only multiplied 1.33 when the cross section area was multiplied 3 in the regenerator. And the average velocity amplitude in the regenerator would only decreased 27.6% at most on the same $|p_i/p_m|$. Furthermore, the local bigger cross section area had a side-effect on the traveling-wave phasing in the regenerator, which limited the extent to extend the regenerator cross section (advised less than treble area from the simulation). According to the simulation, the highest available acoustic power occurred on the 6% of $|p_i/p_m|$, which corresponding mean velocity amplitude, $|\langle u_1 \rangle|$, was 3 m/s. If the area of regenerator was enlarged three times, the highest power occurred on the 2.1 m/s of $|\langle u_1 \rangle|$, as shown in Fig. 9. The most effective operating point occurred on the 4% of $|p_i/p_m|$, which $|\langle u_1 \rangle|$ were 2.25 m/s and 1.6 m/s for the original and the trinal area, respectively. The highest available power efficiency was nearly uniform for different cross section area within the 1–3 times range.

According to Fig. 10, several thermodynamic efficiencies relative to Carnot efficiency varied on different $|p_i/p_m|$ for thermoacoustic Stirling engines. The ideal cycle achieved more and more efficiency gradually with the growth of $|p_i/p_m|$ under 4%. Beyond 4%, the ideal efficiency was increased gently from 70% to 88% (until the 10% of $|p_i/p_m|$), which revealed the perfect thermoacoustic conversion in regenerator similar to the reversible Carnot cycle. If the viscous losses and thermal relaxation losses were considered, the

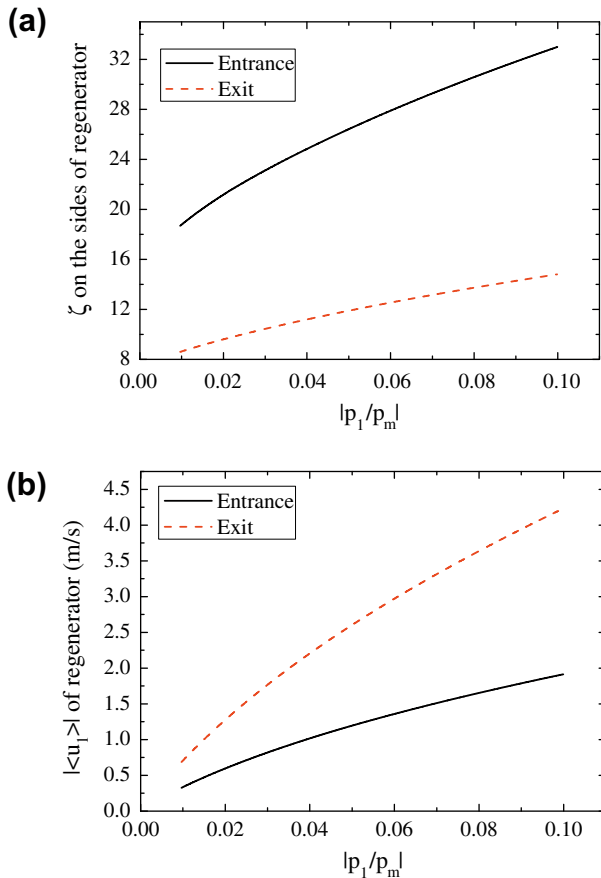


Fig. 6. (a) The dimensionless impedance of the regenerator varied on different pressure ratios. (b) The velocity amplitude of the regenerator varied on different pressure ratios.

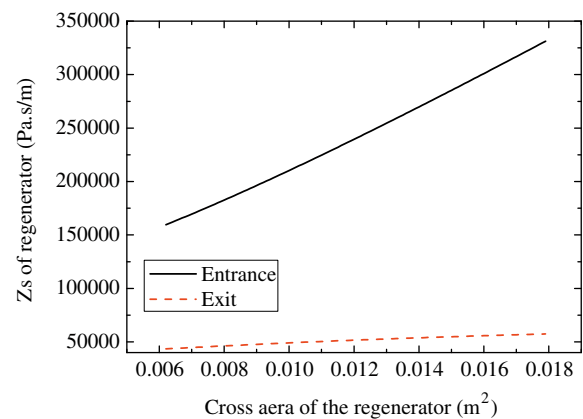


Fig. 7. The impedance for different cross section areas of the regenerator.

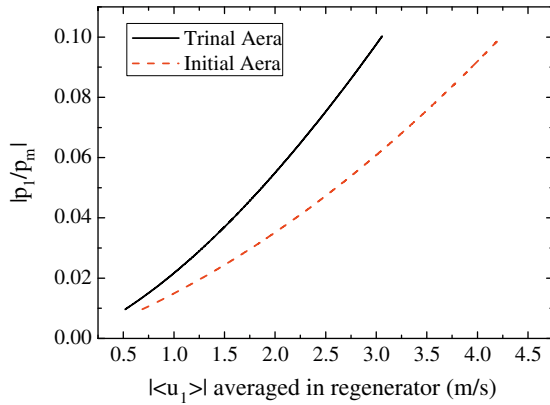


Fig. 8. The linear connection between the average velocity amplitude in regenerator and the pressure ratio with different areas of regenerator.

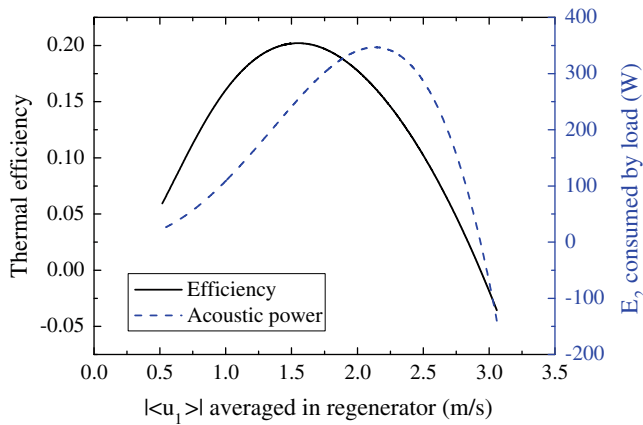


Fig. 9. Compromise between the available power and the efficiency for different velocity amplitudes, when the cross section area of the regenerator was multiplied 3.

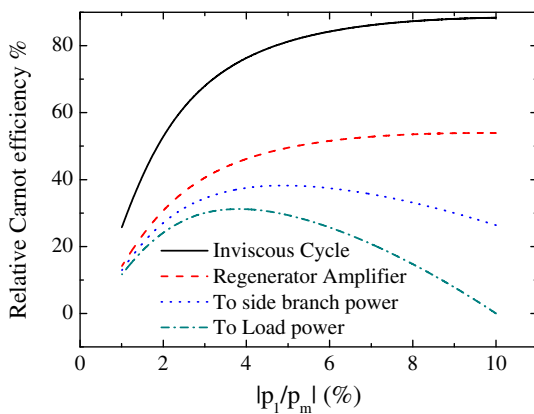


Fig. 10. The relative efficiencies to Carnot efficiency calculated based on different acoustic powers.

varied character of the efficiency in the regenerator was the same as the ideal cycle, except that its values decreased 40% relatively. Nearly 50% of the Carnot efficiency could be achieved in an actual regenerator operating at 10% of $|p_l/p_m|$, which was comparable to that of the common internal combustion engines and piston-driven Stirling engines. But the viscous dissipation occurred in the loop and in the side-branch resonator reduced the capability to ex-

port into loads. Therefore, the thermal efficiency defined by the available acoustic power by load increased first, and then decreased, which peak value was 24.5% corresponding to 32% of the Carnot efficiency operating at 3.8% of $|p_l/p_m|$. According Fig. 11, the comparison between the simulation and the experimental results (experimental data from the Ref. [6]) approved the creditability of the simulation, which most relative error was only 5%. The difference between the available power efficiency and the junction exported efficiency used in literatures was obvious increased with the growth of $|p_l/p_m|$, as was necessary and important to be indicated. The available thermal efficiency would be close to zero but the reported efficiency in literatures might exceed 20% operating on 10% of $|p_l/p_m|$. To show the losses in the regenerator, the dash dotted line in Fig. 11 indicated the losses divided by the heating power, which was also proportional to $|p_l/p_m|$.

Fig. 12 described the percent of the acoustic power consumed by several elements in an actual thermoacoustic Stirling engine. The regenerator power gain was divided into four parts, i.e. the loop frictional loss, the local resistance loss, the side-branch dissipation and the available power consumed by the load. The loop losses included local resistance loss and the frictional loss along the tube, which were both proportional to the $|p_l/p_m|$. And the local losses of loop increased more rapid with the growth of the $|p_l/p_m|$. As the above discussed, $|p_l/p_m|$ had the same meaning as the velocity amplitude. The impedances of the loop were much greater than the side-branch resonator, so the averaged velocity amplitude in the latter would be greater than that in the former. Higher velocity

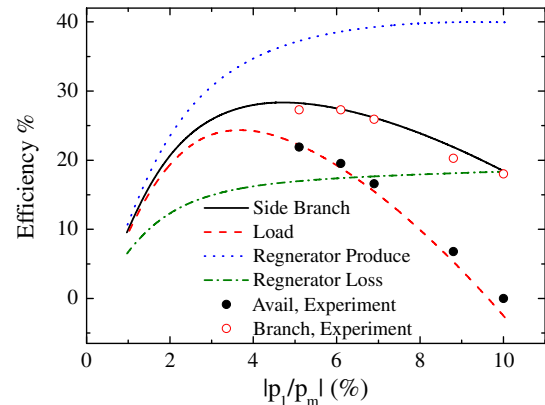


Fig. 11. Comparison between the simulation and the experiments on the thermal efficiencies calculated by different acoustic powers, where the experimental data were got from the Fig. 13 of Ref. [6].

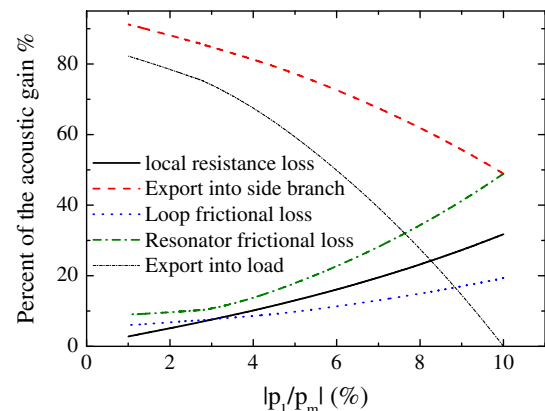


Fig. 12. The proportions of the power gain distributed among the different losses and the available power operating on the different pressure ratios.

meant greater viscous dissipation. Under the 3.8% of $|p_i/p_m|$, the dissipation in the loop had the similar percent of the power gain as that of the side-branch resonator. But beyond the 3.8% of $|p_i/p_m|$, the dissipation in the bypass began to be much more quotient of power gain. So the available power percent of the gain decreased when the $|p_i/p_m|$ grew. It was also shown that the difference between the exported power from the junction and the available power consumed by load increased with the rising $|p_i/p_m|$. On the 7% of $|p_i/p_m|$, half of the exported power from the junction was dissipated in the resonator, and only the 1/3 of the power gain was possible to drive different loads.

On the other hand, the power intensity level changed also with the variety of $|p_i/p_m|$, as shown in Fig. 13. In the considered range of the $|p_i/p_m|$, the available power and the exported power from the junction increased firstly and then decreased like a parabola when $|p_i/p_m|$ increased. But the maximums of the two powers occurred at different $|p_i/p_m|$, 6.5% and 9%, respectively. The difference of the two powers was just the dissipation along the resonator tube, which was proportional to the square of $|p_i/p_m|$. The steep points on the dissipation curve occurred also on 3.8% of $|p_i/p_m|$. Noticed that the peak available power level occurred on 6.5% of $|p_i/p_m|$, but the peak available power efficiency often occurred on 4% of $|p_i/p_m|$. Compromise between the power level and the efficiency level, the operating of $|p_i/p_m|$ should be kept in 4–6.5%, or else the thermoacoustic Stirling engine lost actual significance. It was confirmed that to pursue higher $|p_i/p_m|$ (>6.5%) in thermoacoustic Stirling engines was a fruitless way to develop an effective thermoacoustic technology, according to experiments and calculations.

As mentioned above, the dissipation occurred in the side-branch resonator could be subordinate on the lower $|p_i/p_m|$ (<3.8%), but would be dominant on the higher $|p_i/p_m|$ (>3.8%). The transition of the dissipation resulted from the oscillating flow turning to turbulence. The critical $|p_i/p_m|$ of turbulence was calculated at about 3.76% typically based on the next discussion. The peak available acoustic power occurred at a higher $|p_i/p_m|$ such as 6.5%, because the increase of power gain overran the increase of dissipation under 6.5% of $|p_i/p_m|$. But the decrement would overrun the power gain on the higher $|p_i/p_m|$ (>6.5%). The character of the resonator dissipation depended on the oscillating flow state. The acoustic power per unit length along the resonator could be calculated for the laminar flow and the turbulence using the formula (12) and (13), respectively, where f_M was an approximation to the Moody friction factor, N_{R1} was the peak Reynolds number and ε was the surface roughness [14,19]. Obviously, the dissipation in a resonator would be proportional to the square of volume velocity in laminar flow, but would be proportional to the cube

of volume velocity in turbulence. The acoustic resistance of the resonator would be multiplied $|\langle u_1 \rangle|$ (ordinarily $|\langle u_1 \rangle|$ was much greater than 15 m/s) when the laminar flow turned to turbulence. The decline of the available power efficiency was caused by this non-linear turbulence phenomenon. This non-linear essentiality of turbulence greatly restricted the application on the high $|p_i/p_m|$ (>6.5%).

$$\frac{dE_2}{dx} = \frac{\rho |U_1|^2 \omega}{2\pi r^2} \operatorname{Re} \left[\frac{i}{1 - f_v} \right] \quad (12)$$

$$\frac{dE_2}{dx} = \frac{\rho |U_1|^3 \omega}{3\pi^3 r^5} \operatorname{Re} \left[f_M - (1 - 9\pi/32) N_{R1} \frac{df_M}{dN_{R1}} \right] \quad (13)$$

$$\frac{1}{\sqrt{f_M}} = 1.74 - 2 \log_{10} \left(2\varepsilon + \frac{18.7}{N_{R1} \sqrt{f_M}} \right)$$

The acoustic dissipation in the side-branch resonator could be calculated using boundary approximation, because the radius of the flow channel was much larger than the penetration depths. This dissipation was caused by the viscous friction where close to the solid boundary. The critical Reynolds number to judge turbulence referred to formula (14), where N_{R1} was defined as the peak Reynolds number expressed in formula (15) [14]. According to formula (12), the acoustic dissipation per unit inner area could also be written as formula (16). The viscous dissipation term showed the viscous shearing at an average rate angle frequency, which denoted serious dissipation on the high $|p_i/p_m|$. For the mentioned thermoacoustic Stirling engine, the critical peak velocity amplitude for turbulence in the side-branch resonator was 17.4 m/s, corresponding to a 3.76% of $|p_i/p_m|$. The critical peak-to-mean pressure ratio of turbulence was just near that of the maximum available power efficiency.

$$\frac{N_{R1}}{D/\delta_v} \geq 500 \quad (14)$$

$$N_{R1} = \frac{4\langle u_1 \rangle r_h}{v} \quad (15)$$

$$\frac{dE_{2\text{loss}}}{ds} = -\frac{1}{4} |\langle u_1 \rangle|^2 \delta_v \omega - \frac{1}{4} \frac{|p_i|^2}{\gamma p_m} (\gamma - 1) \delta_v \omega \quad (16)$$

The above conclusions and the advised $|p_i/p_m|$ were propagable for all the thermoacoustic Stirling engines with different sizes. But the different working fluids had some effects on the critical peak-to-mean pressure ratio that caused the most efficiency or the highest power level. For example, the highest $|p_i/p_m|$ was advised on 6.5% for helium and 7.75% for nitrogen, respectively. Because the velocity of sound of nitrogen was 36% of that of helium, and the specific acoustic impedance of nitrogen was 2.44 times of that of helium, the $|p_i/p_m|$ in nitrogen would be 2.33 times of that in helium at the same velocity amplitude. Considered other difference in physical performance between the nitrogen and helium, the penetration depths of helium were about 1.7 times of that of nitrogen at the same device. The critical N_{R1} in nitrogen was about 4.55 times of that in helium. So the velocity amplitude using nitrogen would be 0.51 of that using helium if the same ratio of the available power to the power gain was kept. It was easy to calculated that the most effective and the maximum power level using nitrogen both occurred on the 1.19 times $|p_i/p_m|$ of that using helium. Other working fluids, such as argon and carbon dioxide, etc., could also be discussed similarly, which was used rarely in thermoacoustic engines due to unfit freezing point or great viscosity.

4. Contrast between traveling-wave engine and standing-wave engine

Compared to thermoacoustic Stirling engines, the standing-wave thermoacoustic engines had been developed in the recent

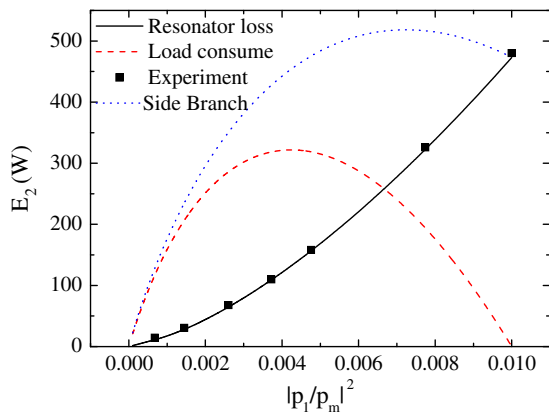


Fig. 13. The dissipated acoustic power in resonator and the available power of the load on different pressure ratios, which experimental data came from the Fig. 11 of Ref. [6].

30 years that was the simplest type of thermoacoustic engines [18,20]. The thermodynamic cycle in the standing-wave thermoacoustic engine was similar to the Brayton cycle. Because the inherent irreversible heat transfer loss restricted its thermal efficiency, only about 18% of thermal efficiency had been achieved in the best standing-wave thermoacoustic prototype engine built by Tektronix Company [21]. This reported efficiency was calculated using the available power to drive a pulse tube refrigerator, which was about 23% of Carnot efficiency. Hence, the available power efficiency relative to Carnot efficiency for the reported standing-wave thermoacoustic engine was 76% of the highest efficiency for the best thermoacoustic Stirling engine. But the power density of the standing-wave engine was about 3.7 times of that of the thermoacoustic Stirling engine. And the effective operating peak-to-mean pressure ratio of pressure amplitude was about 1.7 times of that in the thermoacoustic Stirling engine. Besides, there were other ineluctable non-linear phenomena that reduced thermoacoustic Stirling engines' efficiency, such as the Gedeon streaming in the traveling-wave loop, the frequency transition, the high-order harmonics and the thermal stress question in the loop, and so on [14]. Especially the Gedeon streaming, the net time-averaged mass flow along the loop circuit caused by the second-order pressure gradient, would change the temperature distribution and arose serious heat leak between the hot and cold heat exchangers [22]. To suppress Gedeon streaming, some methods had been approved ineffective, such as the "jet pump" using the asymmetry of pressure gradient. Jet pump would consume more than 15% of power gain to suppress Gedeon streaming, and it was difficult to tune on different operating conditions. Gedeon streaming was also a restricted factor in the application of the thermoacoustic Stirling engines. Hence the standing-wave engine was more competitive than the thermoacoustic Stirling engine.

Although the thermodynamic cycles of the traveling-wave thermoacoustic engine was more perfect than that of the standing-wave thermoacoustic engine, the viscous losses of the former was much greater than that of the latter. Because the hydraulic radius of the channels in the regenerator of the standing-wave engine was about 1.2 times of the thermal penetration depth, but the hydraulic radius for the traveling-wave regenerator channels should be 0.1 of the thermal penetration depth. If the phasing p_1 leading U_1 was tuned at about 70–85°, and the impedance was also enhanced, the quasi-standing-wave thermoacoustic engine could achieved more than 30% of Carnot efficiency, as was concluded from the investigation on cascade thermoacoustic engines [10,11]. The length of the regenerator in the standing-wave thermoacoustic engines was often five times than that in the traveling-wave thermoacoustic engines, which would decrease thermal conduction deeply along the metal wall between different temperatures. Longer regenerator meant more particles attended thermodynamic cycle to produce acoustic power, so the power intensity was larger than that of the shorter regenerator at the same $|p_1/p_m|$. According to the previous study on the standing-wave thermoacoustic engine, 40% of the power gain was dissipated in heat exchangers and in the resonator operating on the 10% of $|p_1/p_m|$. Tektronix Company had decreased this dissipation percent to 28% of the power gain [18–21]. Because relative less power density, the dissipation in the traveling-wave engine looked more significant, which dissipation often exceeded 70% of power gain operating on the 10% of $|p_1/p_m|$.

The simulations of the standing-wave thermoacoustic engine built in Tektronix Company, operating on 350 Hz, approved that the efficiency, power density, and peak-to-mean pressure ratio could be achieved higher levels. It would be super in the occasion that required high intensity and high pressure ratio. On the 10% of $|p_1/p_m|$, the traveling-wave engines demonstrated its efficiency would attain more than 45% higher than that of the standing-wave

engines. But the available power efficiency for standing-wave engines was more 2.5 times than that of the traveling-wave engines. And the available power intensity was enhanced 13 times than the traveling-wave engine. Even at the most effective operation condition, the available power density of the traveling-wave engines was only a quarter of that of the standing-wave engines.

According to the classic thermoacoustic theory, some simulation results were shown in Figs. 14–16. Fig. 14 showed that the

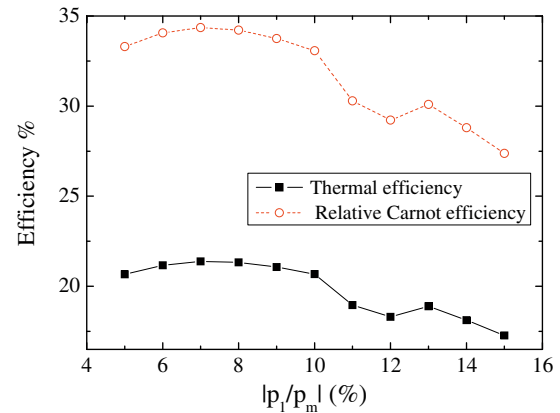


Fig. 14. The thermal efficiencies on different operating pressure ratios were calculated in the standing-wave engine built in Tektronix Company.

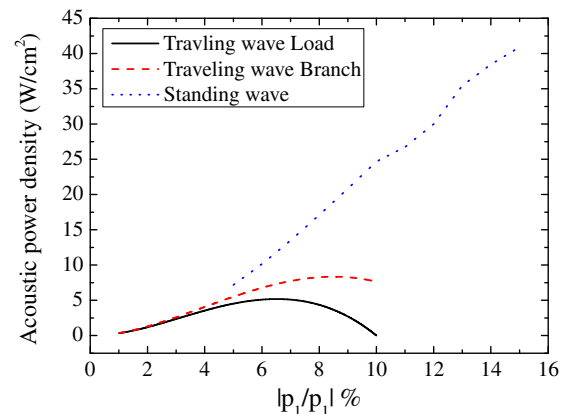


Fig. 15. The comparison of the acoustic intensity for the traveling-wave engine and the standing-wave engine, based on simulations.

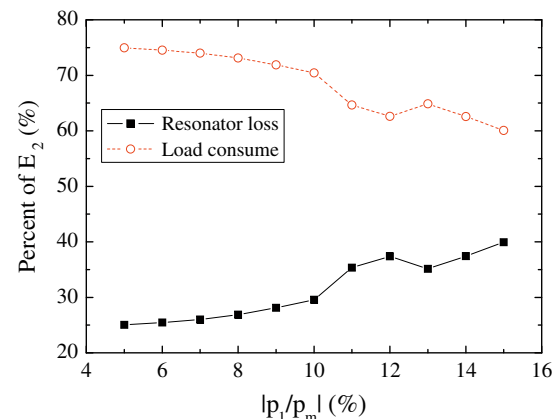


Fig. 16. The distributed proportion of the power gain between resonator and load for the standing-wave engine.

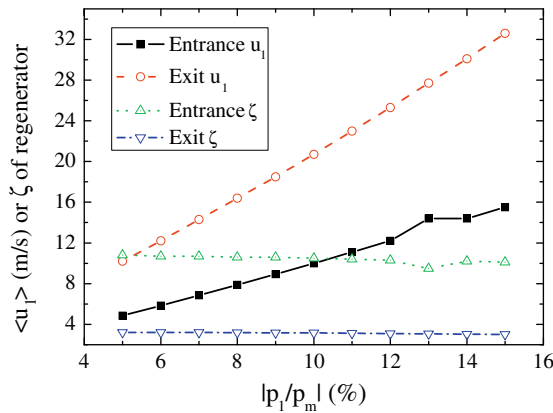


Fig. 17. The velocity amplitudes and the dimensionless impedance of the regenerator varied on different pressure ratios for the standing-wave engine.

thermal efficiency of standing-wave engines would also be reduced gradually when the operating pressure ratio increased, which degradation was slower than that occurred in thermoacoustic Stirling engines even on the high pressure ratio. Fig. 15 compared the growths of power intensity with the rising pressure ratio between the standing-wave engine and the traveling-wave engine. Fig. 15 indicated the better character of the standing-wave engines. The proportion distributed between the resonator dissipation and the available power in the standing-wave engine was also shown in Fig. 16, which dissipation was much lesser than the available power on the higher pressure ratio. The high acoustic intensity came from the high velocity amplitude in the regenerator for the standing-wave engines, as shown in Fig. 17, but the impedance had minor changes on different pressure ratios.

5. Improving direction of thermoacoustic engines

As the evaluation mentioned, some advices to improve the thermoacoustic Stirling engines were provided. As a simple idea, turbulence might be weakened by the variation of the resonator diameters at different volume velocities, which was often adopted using a cone or a clarion as the side-branch resonator. But the cone resonator could not restrain turbulence on higher pressure ratio, for example the so-called energy-focused resonator had still dissipated more than 73% of the power gain [9]. To get more effective improving methods, some novel ways were advised as below.

Firstly, tuning the phasing conditions between the standing-wave and the traveling-wave, called a generic wave or quasi-standing-wave, could enjoy the advantages of the standing-wave engines and the traveling-wave engines. It was in nature the compromise between the irreversible losses and the viscous losses in regenerators, which would achieve more power density on a higher available power efficiency.

Secondly, the relative proportion of thermoacoustic cores was being increased when multi-stage thermoacoustic cores were cascaded [23]. The acoustic power was created through different regenerators, in which the velocity amplitude could be decreased even to achieve higher acoustic intensity.

Thirdly, compact thermoacoustic engines were possible to be developed in high frequency even in ultrasonic frequency [24]. For example, the high frequency standing-wave engines reduced the assistant resonator size with the longer regenerator than the thermoacoustic Stirling engines', as enhanced the available power efficiency and intensity. Using solid as the working medium operating in ultrasonic frequency, the acoustic intensity should be enormous. The traveling-wave could also be tuned by the coupling

of dual ranges of standing-wave, similar to the ultrasonic motors [25].

Fourthly, the essential method was to eliminate the side-branch resonator of thermoacoustic Stirling engines, which might be replaced by a piston-spring such as the generators. But the power intensity had not been increased greatly due to turbulence occurred in the loop circuit [26]. The impedance matching conditions for the generator required to be controlled strictly, which was often difficult. So the side-branch resonator was difficult to eliminate [27].

Fifthly, the cone regenerator could also decrease viscous losses and extended an effective traveling-wave region length. But it was difficult to realize in manufacture.

Lastly, the mixture working gas with lower Prandtl number could reduce the viscous domain in flow channels, because the ratio of the viscous penetration depth to the thermal penetration depth was proportional to the square root of Prandtl number. According the Wilke-Wassiljewa calculation method, the mixture of helium and xenon could achieve as low as 0.2 of the Prandtl number, which viscous penetration depth would be reduced by half [28].

6. Conclusions

Based on the above discussion and simulation analysis on the evaluation of the thermoacoustic Stirling engine, some important viewpoints were concluded as below.

- (1) The thermodynamic cycle in the regenerator of the thermoacoustic Stirling engines was very close to the Stirling cycle, with a thermal-to-PV power conversion efficiency of up to 30%. It revealed thermoacoustic engines might be competitive and comparable to the internal combustion engines.
- (2) There was an error on the efficiency evaluation in previous literatures, because the reported thermal efficiency based on the power exported into the bypass, included the inevitable viscous dissipation of the side-branch resonator that was an essential assistant configuration. Operating on the higher pressure ratio ($|p_1/p_m| > 6.5\%$), more than 70% of acoustic power gain would be consumed by the side-branch resonator. The dissipation was the transfer of the viscous dissipation that should occur in the regenerator if the bypass was eliminated. The available power thermal efficiency was often under 20%, as restricted the application on higher pressure ratio conditions.
- (3) The laminar flow and turbulence induced the different available power efficiency, which flow state depended on the peak-to-mean pressure ratio. Turbulence occurred on the high pressure ratio (usually $|p_1/p_m| > 4\%$) and dissipated most of the exported acoustic power from the junction. Turbulence was a non-linear essence of the viscous resistance in resonators.
- (4) The effective thermoacoustic Stirling engine operated on the enough low peak-to-mean pressure ratio (4–6.5%), which achieved about 30% of Carnot efficiency. But the available power density would under 10 W/cm^2 on this high efficiency level.
- (5) Introducing the moving parts into thermoacoustic Stirling engine, such as the piston-string of a generator to eliminate the side-branch resonator, could take advantage of its perfect thermodynamic cycle. However, the thermoacoustic generator had the limited power density and efficiency on the high pressure ratios.
- (6) According to the actual thermal efficiency, the available power density and the pressure ratio, the standing-wave

thermoacoustic engines would outvie the traveling-wave thermoacoustic Stirling engines.

- (7) The multi-stage thermoacoustic engines and ultrasonic thermoacoustic engines were both the important improving ways to develop an applied thermoacoustic technology.

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